Design of Spur Gear

19.04.2019

Aim: (i) To determine dimensions of gear tooth [i.e. $D = mz$; $b = \psi_m = 10m$, $P_c = \pi m$; add. $= m$; ded $= 1.157m$)

(ii) Module is determined by using beam strength equation of gear tooth.

(iii) Checking the dimensions of the gear tooth w.r.t. bending and wear failure (i.e. $F_{dynamic} \le F_{beam\ strength}$ & $F_{dynamic} \le F_{wear\ strength}$)

★ Force Analysis:

For two mating or meshing gears:

(i) $\phi_1 = \phi_2 = \phi$ (Pressure angle is same)

(ii) $P_1 = P_2 = P$ (Power is same, $\eta_{mech} = 100\%$)

(iii) $F_{t_1} = F_{t_2} = F_t$

(iv) $m_1 = m_2 = m$.

(v) $F_{r_1} = F_{r_2} = F_r$.

(vi) $(F_R \text{ or } F_n) = \sqrt{F_t^2 + F_r^2}$ $\therefore F_{R_1} = F_{R_2} = F_R$.

(vii) $Z_1 \neq Z_2$.

(viii) $D_1 \neq D_2$.

(ix) $N_1 \neq N_2$

(x). $T_1 \neq T_2$.

Load / M/C component	F_r	F_t
Gear tooth	A.C.L.	T.S.L
Shaft	T.S.L.	E.T.S.L $(e=R)$ = TSL + $(TM = F_t \times R)$.

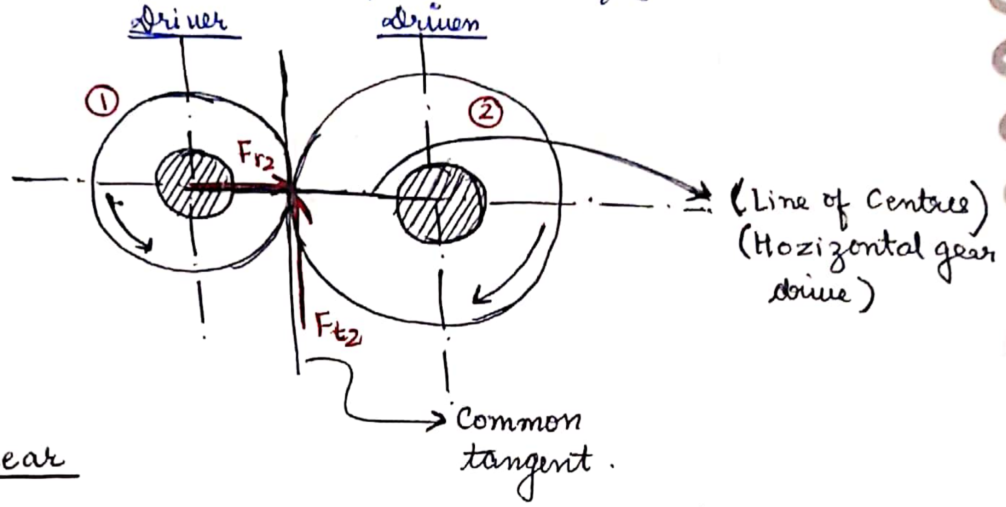
(i) Torque $(T) = \dfrac{60 \times 10^6 \times P}{2\pi N} =$ ________ Nmm.

(ii) $F_t = \dfrac{2T}{D}$

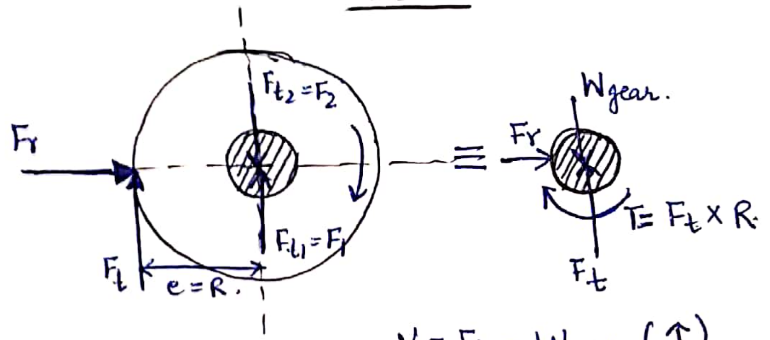
(iii) $F_r = F_t \tan\phi$.

(iv) F_R or $F_n = \sqrt{F_t^2 + F_r^2}$

* F_r will be always along the line of centre.
* F_t will be along the common tangent.
* W.r.t. gear, the direction of F_r will in the direction of power transmission
* Direction of F_t depends on direction of rotation of gear.



Gear

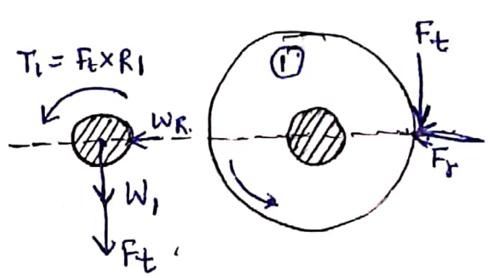


$V = F_t - W_{gear}$ (↑)

$H = F_r$ (→)

$T = F_t \times R$ (↻)

Pinion



$V = F_t + W_1$ (↓)

$H = F_r$ (←)

7 diagrams is drawn:

* For TSL : One SFD, 1 BMD.
* For ETSL : 1 SFD, 1 BMD
* Horizontal and Vertical Resultant diagrams (SFD, BMD)
* Torque diagram.

Out of the above 7 diagrams, SFD in both cases is not drawn as shear stress on extreme fibres are zero and BM and TM are going to be maximum.

∴ Finally, there are diagrams

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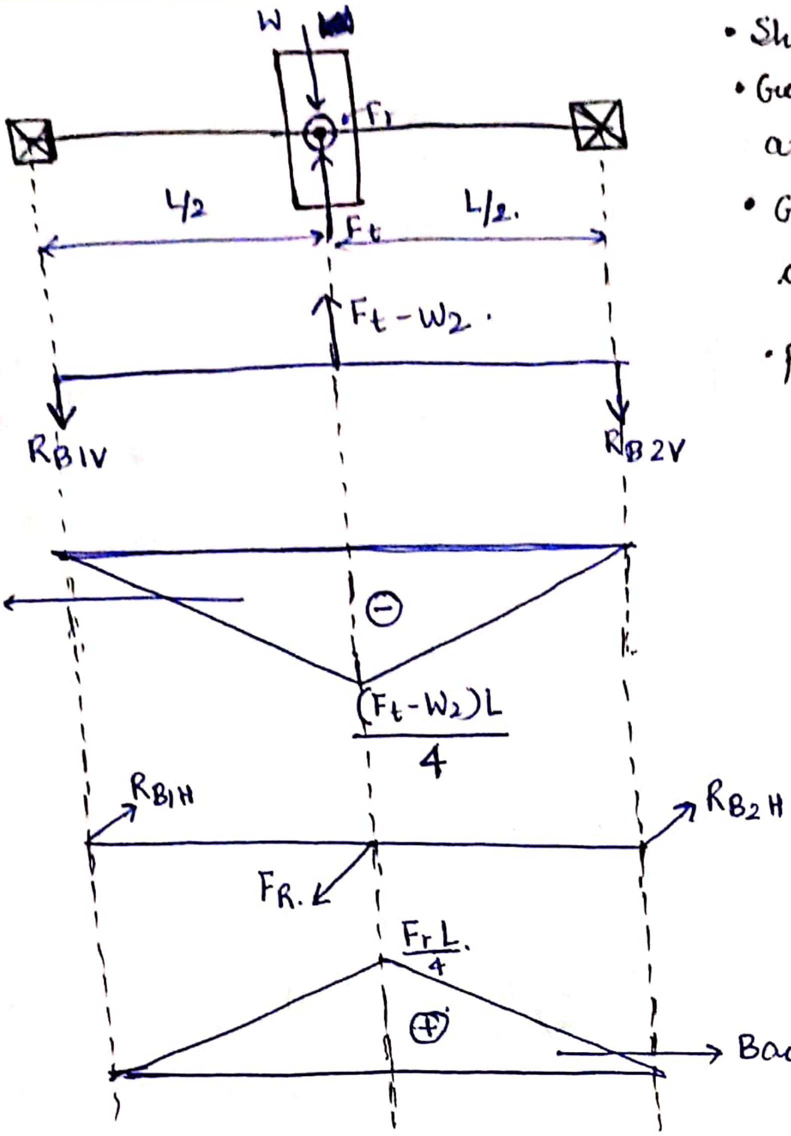
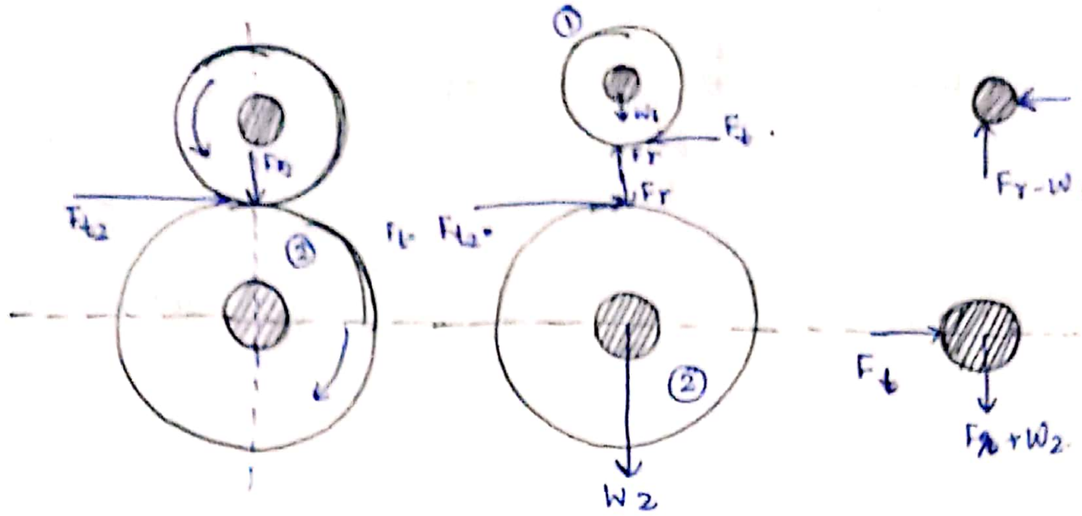
• Torque diagram.



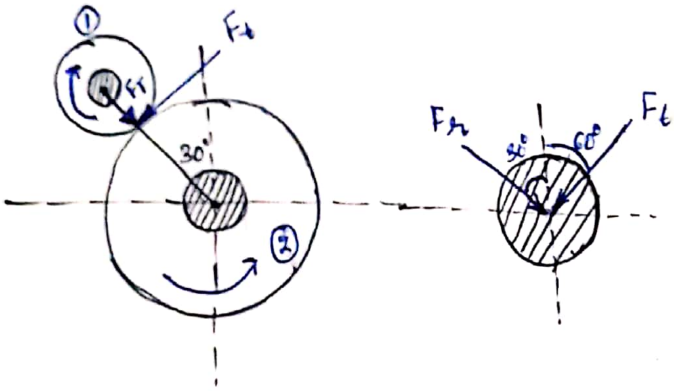
Bottom surface gets contracted

Back side contracts.

- Short and square bearings
- Gear is mounted exactly at mid span.
- Gear rotation direction depends on F_t
- Pinion rotation direction is decided by the driver shaft. It does not depends on F_t.



Vertical gear drive

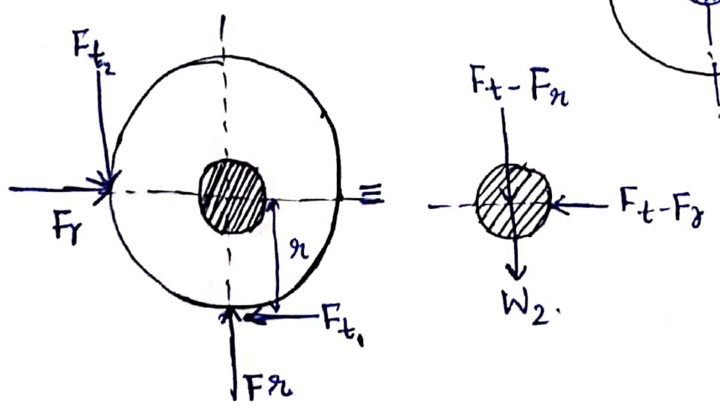


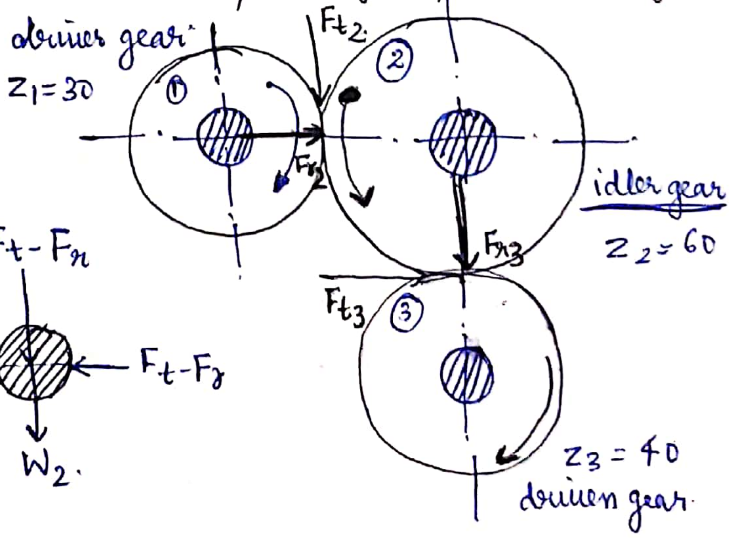
$H = F_r \sin 30 - F_t \sin 60°$

$V = F_r \cos 30° + F_t \cos 60° + W_2$

Q For the gear train as shown. Determine resultant force on idler gear shaft. Assume $P = 3.5\,kW$ at 700 rpm. in CW. direction. module, $m = 5mm$. no. of teeth on ~~pinion~~ driver gear, idler gear and driven gear are, 30, 60 and 40 respectively, pressure angle is 20°

idler gear :





Net torque is zero $\left[(F_{t_1} \times r)_{ACW} + (F_{t_2} \times r)_{CW}\right]$ hence, it's called idlers

$F_{t_1} = F_{t_2} = F_t$.

F_{t_2} & $F_t \Rightarrow T_1 = F_t \times R$ (CW)

$F_{t_1} = F_t \Rightarrow$ HTSL ($\leftarrow$)

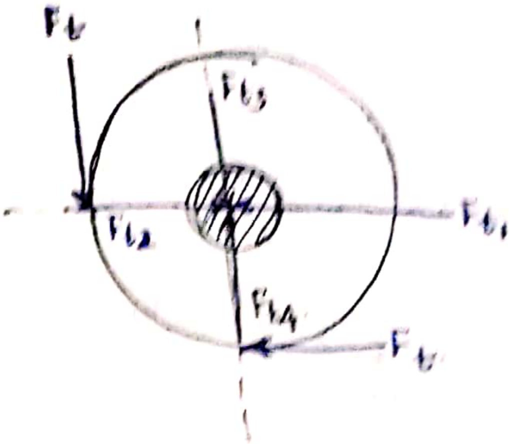
$F_{t_3} = F_{t_4} = F_t$.

F_t and $F_{t_4} \Rightarrow T_2 = F_t \times R$ (ACW).

$F_{t_3} = F_t \Rightarrow$ VTSL ($\downarrow$)

$\therefore T = T_1 + T_2 = 0$

$P_2 = T\omega = 0$.



$\therefore$ Idler shaft is designed either ~~bes~~ using bending equation or strength criterion. [theories of failures are optional]

* For idler gear shaft:

$H = F_t - F_r$ ($\leftarrow$)

$V = F_t + W_2 - F_r$ ($\downarrow$) $= F_t - F_r$ ($\downarrow$) neglecting weight

$R = \sqrt{H^2 + V^2} = \sqrt{2}\,H = \sqrt{2}\,V = \sqrt{2}\,(F_t - F_r)$

① $T_1 = \dfrac{60 \times 10^6 P}{2\pi N_1} = 47746.48293$ N-mm.

② $T_2 = 0$

③ $T_3 = \dfrac{60 \times 10^6 P}{2\pi N_3}$ or $\dfrac{T_3}{T_1} = \dfrac{Z_3}{Z_1}$ (for $\eta_m = 100\%$) $T_3 = 63661.97724$ N-mm

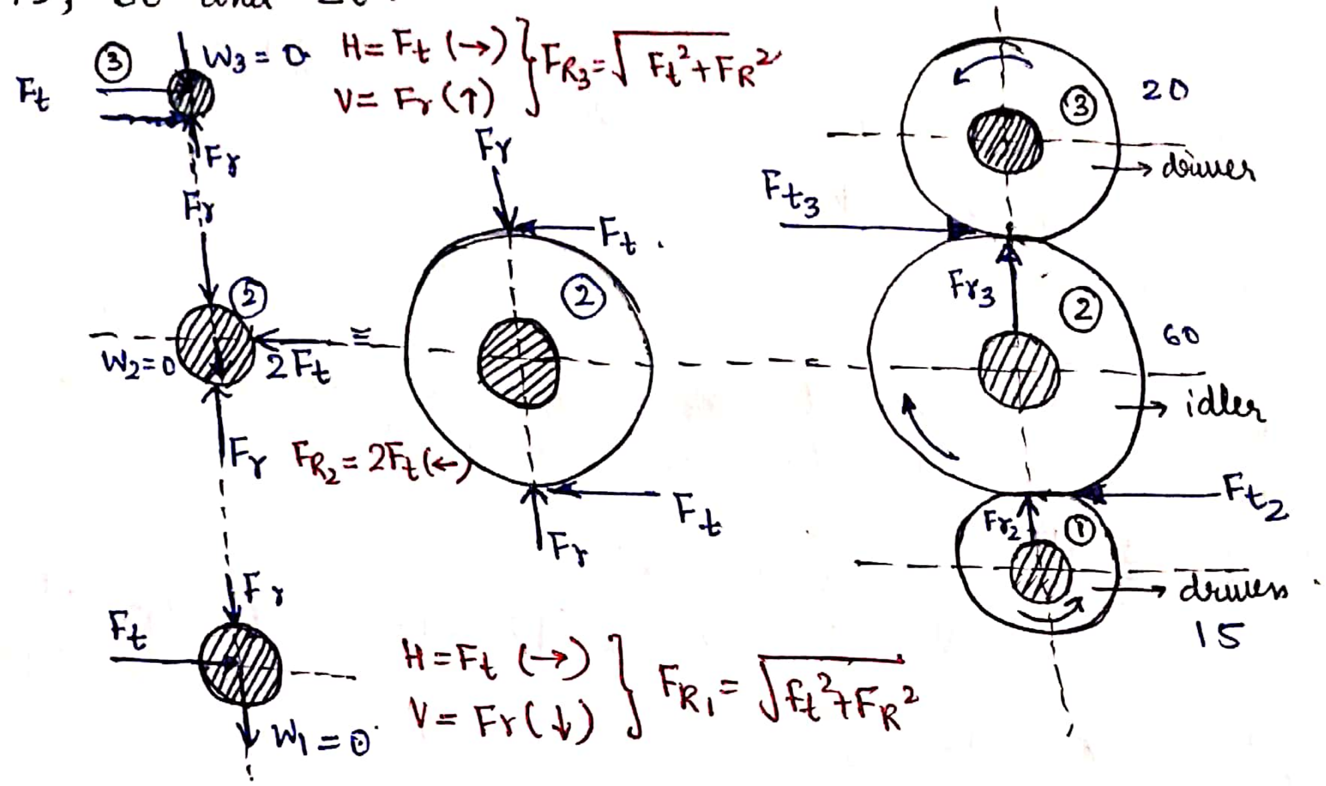
④ $F_t = \dfrac{2T_1}{D_1}$ or $\dfrac{2T_3}{D_3} = 636.6198$ N.

⑤ $F_r = F_t \tan\phi = 231.7106$ N.

⑥ $(F_R)_2 = \sqrt{2}\,(F_t - F_r) = 572.628$ N.

⑦ $F_{R_1} = F_{R_3} = \sqrt{F_t^2 + F_r^2} = 677.47$ N. (by neglecting weights)

Q) For gear train as shown in Fig. determine resultant forces on driver shaft, idler gear shaft and driven shaft. P = 10 kW at 500 rpm module = 5mm. ; $\phi = 20^\circ$; no. of teeth on driver, idler and driven are 15, 60 and 20.



1. $T_1 = \dfrac{60 \times 10^6 \times P_0}{2\pi N_1} = 190985.9317$ N-mm.

2. $T_2 = 0$

3. $\dfrac{T_3}{T_1} = \dfrac{z_3}{z_1} = 254647.9089$ N-mm

4. $F_t = \dfrac{2T_1}{D_1}$ or $\dfrac{2T_3}{D_3} = 5092.958$ N.

5. $F_r = F_t \tan\phi = 1853.685$ N.

6. $F_{R_1} = F_{R_3} = \sqrt{F_t^2 + F_R^2} = 5419.812$ N.

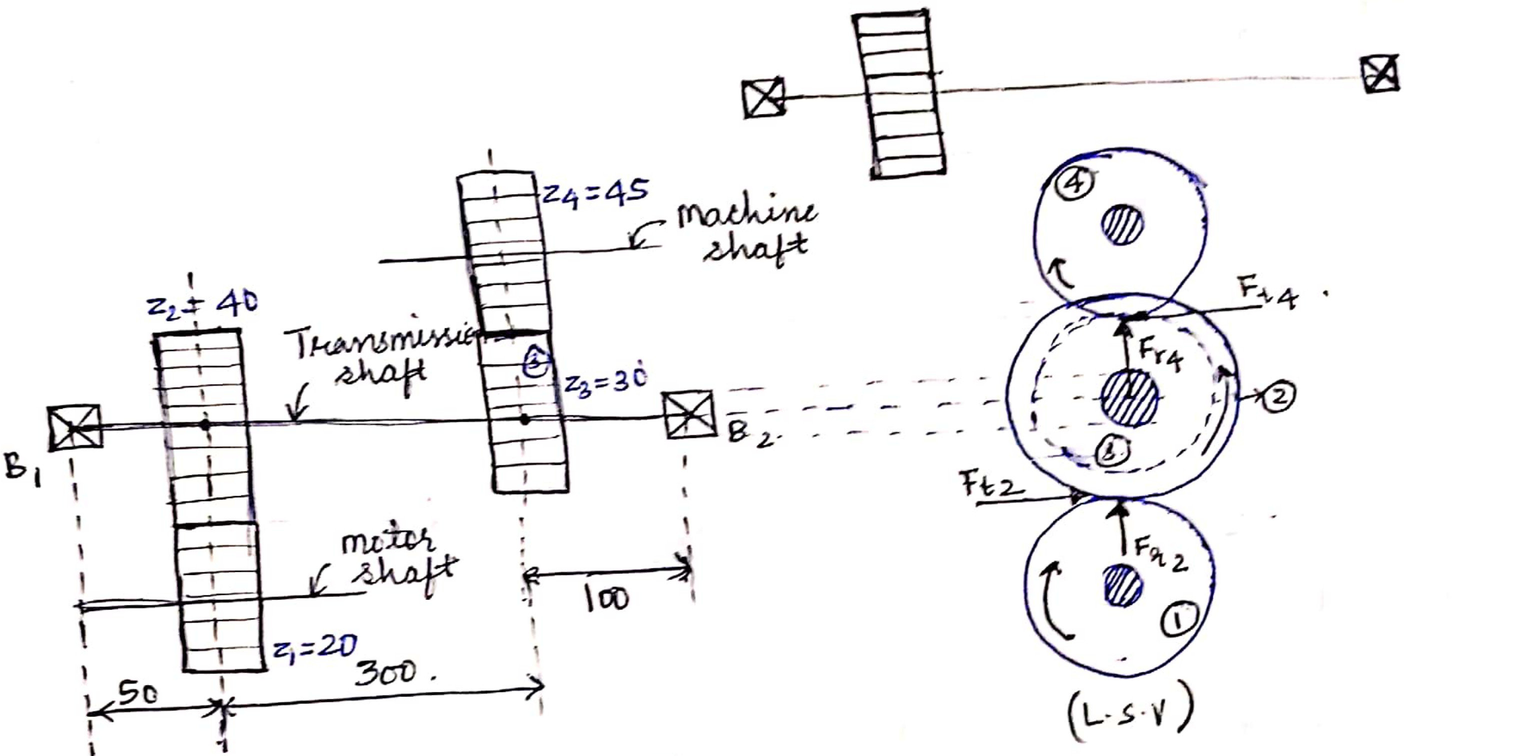
7. $F_{R_2} = 2F_t = 10185.916$ N.

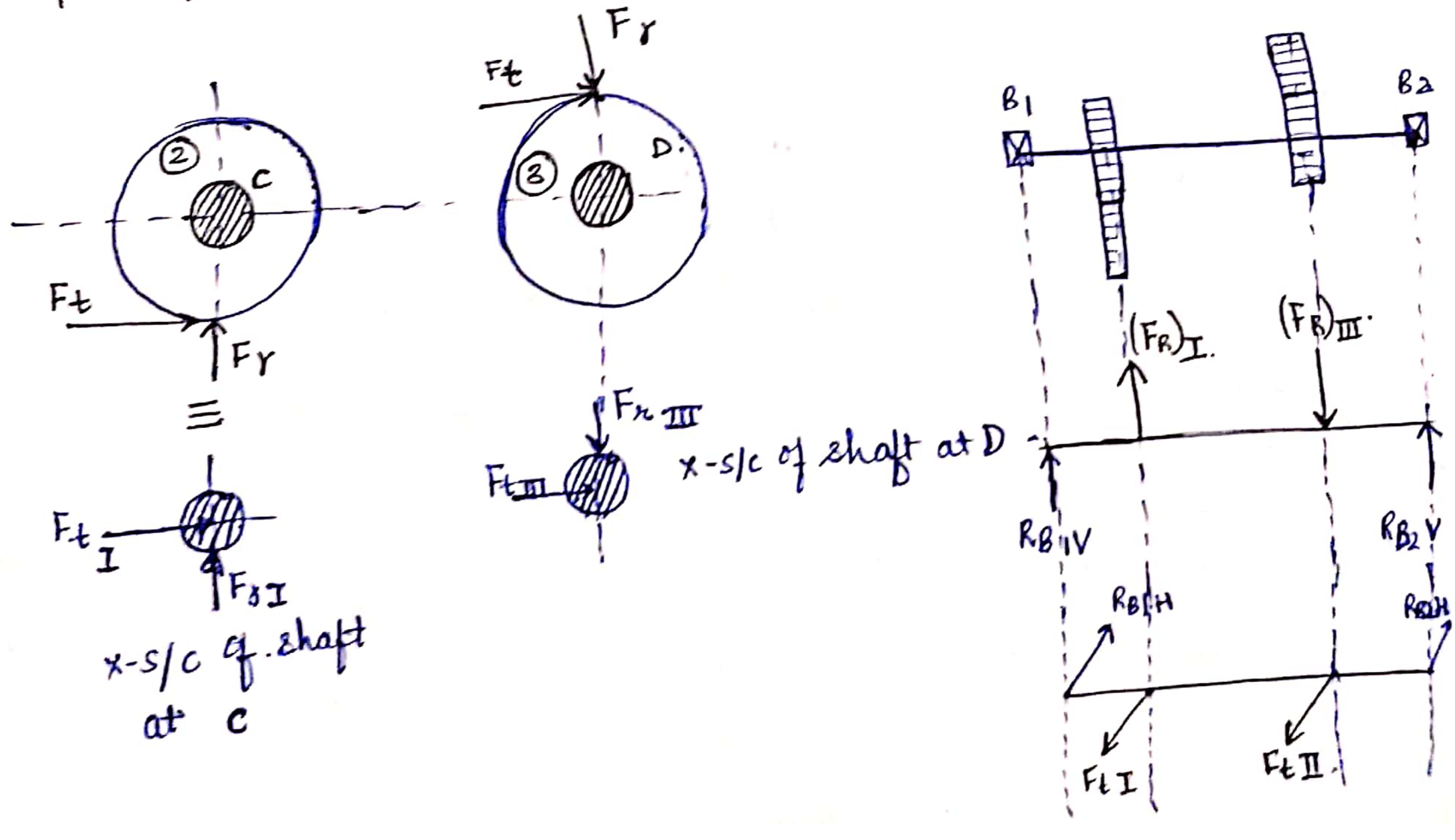
Q Determine Reactions on bearing supporting the shaft upon which gears 2 and 3 are mounted as shown in Fig. Gear 1 is driver with 20 teeth, 6mm module, 20° full depth gear, rotating at 2000 rpm in CW direction and transmitting 80 kW.

Gear 2 : 40 teeth   Gear 3: 5mm module 20° F.D. with 30 teeth.

Gear 4 has 45 teeth

Soln.





$$T_1 = \frac{60\times10^6 \times f}{2\pi N_1} = 381.9718 \times 10^3 \text{ N-mm.}$$

$$(F_t)_I = \frac{2T_1}{D_1} = \frac{2T_1}{20\times6} = 6366.1977 \text{ N}$$

$$(F_r)_I = (F_t)_I \tan\phi = 2317.1064 \text{ N.}$$

$$\frac{T_2}{T_1} = \frac{Z_2}{Z_1} \Rightarrow T_2 = 763943.6 \text{ N-mm} = T_3 . \quad \left(\begin{matrix} P_3 = P_2 \\ N_3 = N_2 \end{matrix}\right)$$

$$(F_t)_{III} = \frac{2T_3}{D_3} = 10185.9147 \text{ N.}$$

$$(F_r)_{III} = (F_t)_{III} \tan\phi_{II} = 3707.3697 \text{ N.}$$

$$F_{t1} = F_{t2} = F_{t\,I} \quad ; \quad F_{r1} = F_{r2} = F_{r\,I}$$

$$F_{t3} = F_{t4} = F_{t\,III} \qquad F_{r3} = F_{r4} = F_{r\,III}$$

$$R_{B_1V} = \frac{(-2317.106)(400) + (3707.36)\times100}{450} = -1235.792 \text{ N.}$$

or $R_{B_1V} = 1235.792$ N $(\downarrow)$

$$R_{B2V} = +1235.792 - 2317.106 + 3707.36 = 2626.046 \text{ N.}$$

$$R_{B1H} = \frac{(6366.1977\times400) + (10185.89\times100)}{450} = 7922.373 \text{ N} (\nearrow)$$

$$R_{B2H} = 8629.71 (\nearrow)$$

∴ <u>Resultant reaction on bearing:</u>

$$R_{B1R} = \sqrt{(R_{B1V})^2 + (R_{B1H})^2} = 8018.176 \text{ N.}$$

$$R_{B2R} = \sqrt{(R_{B_2V})^2 + (R_{B_2H}^2)} = 9020.4 \text{ N.}$$

* Journal bearing is used if speed is higher and shaft works continuously
* (Cylindrical roller) bearing ⟨Anti-friction⟩ if low speed and intermittent operation
* Transmission shaft → • line shaft (used when more than ~~two~~ one machines are to operate with single motor)
  • counter shaft (here in this case, the transmission shaft is counter one)
  ↓
  used when one machine is to operate [own 2 gear drives]

Shafts

- **M/C SHAFTS** → integral with driver and driven m/c
  - eg: motor shaft, I.C engine crank shaft.
- **TRANSMISSION SHAFT** → intermediate shafts which are used to receive power from driven m/c & deliver the power driven m/c.
  - **LINE SHAFT** → Used to distribute power among various m/c
  - **COUNTER SHAFT** → Used to obtain higher speed reduction.

★ Design procedure used in spur gears:-

Input Data:

1. $P = x$ kW at Y rpm.
2. Pressure angle $(\phi) =$ ____
3. Gear ratio $(G) =$ ____ .
4. $[\sigma_b] =$ per bending stress $= \dfrac{\text{failure stress}}{N}$. $(N = 3$ default$)$.
5. Lewis form factor$(y) =$ ______
6. $C_v =$ Velocity factor $=$ ______.
7. $\sigma_{es} =$ surface endurance limit $= (2.8 \times BHN - 70)$ MPa.
8. $E =$ young's modulus.

Optional:

9. No. of teeth on pinion (z_1) (by using Rack & Pinion formula)
10. $\psi = b/m = 8$ to 14 (default $=10$)
11. Service factor $(C_s) = 1.2$ to 1.3 default.

Steps: Subscripts 1 and 2 for pinion and gear respectively -

1) $T_1 =$ torque to be transmitted by pinion.

(Mean torque) $T_1 = \dfrac{P \times 60 \times 10^6}{2\pi N_1} =$ ________ N-mm.

2) $[T_1] =$ Design torque for pinion $= T_1 \times C_s$ (max. torque)

3) $z_1 \geq (z_1)_{min}$. where $(z_1)_{min} = \frac{2A_W}{\sin^2\phi}$ for F.D; $A_W = 1$.
for stub tooth, $A_W = 0.8$

4) $z_2 = G\, z_1$

5) module (m) :- module is determined from following eqn which is obtained from beam strength of weaker gear (W.G.)

$$m \geq 1.26 \sqrt[3]{\frac{[T_1]}{([\sigma_b]Y)_{W.G.}\ \psi z_1}}$$

** when gear and pinion are made of same material. pinion is the weaker gear.

$[\because [\sigma_{b_1}] = [\sigma_b]_2$; $b_1 = b_2 = b$; $m_1 = m_2 = m$;

$Y_1 < Y_2 \Rightarrow [F_s]_1 < [F_s]_2$.

Hence; $([\sigma_b]Y)_{W.G.} = [\sigma_b]_1 Y$.

** when they are made of different materials.;

$([\sigma_b]Y)_{W.G} = \min$ of $[\,[\sigma_{b_1}]Y_1$ and $[\sigma_{b_2}]Y_2\,]$.

$Y = \pi y$.

$\therefore m \geq$ ________ mm.

6) dimensions of Gear train :-

$D_1 = m z_1 =$ ________ mm.

$D_2 = m z_2 =$ ________ mm.

$b = \psi m = 10m =$ ________ mm.

7) F_s = beam strength of W.G.

$F_s = ([\sigma_b]Y)_{W.G}\, b\, m =$ ________ N.

8) F_d = dynamic load.

I Method:

(i) $F_t = \frac{2[T_1]}{D_1}$ or $\frac{2[T_2]}{D_2}$.

(ii) $V_1 = V_2 = V = \frac{\pi D_1 N_1}{60}$ or $\frac{\pi D_2 N_2}{60} =$ ________ m/s

iii) $C_v = \frac{3+V}{3}$ or $\frac{3}{3+V}$ (if $V \le 10 m/s$).

(iv) $F_d = F_t \times C_v$ or $\frac{F_t}{C_v}$ = ______ N.
$\hookrightarrow C_v > 1$; $C_v \rightarrow C_v < 1$.

if $F_d \le F_s \Rightarrow$ design is safe w.r.t. bending failure.

II method: (Buckingham eq^n):

$$F_d = F_t + F_i \leftarrow \text{incremental load.}$$

$$F_i = \frac{21 V (Cb + F_t)}{21V + \sqrt{Cb + F_t}}$$, where C = dynamic load const in N/mm.

$$C = \frac{Ke}{\left[\frac{1}{E_1} + \frac{1}{E_2}\right]} \leftarrow mm.$$

e = error in tooth action in mm.
$K = 0.11$ for $\phi = 14\frac{1}{2}°$
$k = 0.114$ for $\phi = 20°$

9) Wear strength (for pinion only)

$F_W = D_1 Q k b$ in N., where $Q = \frac{2G}{G \pm 1}$ (+) for external gears, (−) for internal gears

* $\boxed{F_d \le F_W}$ Design is safe w.r.t. wear failure.

k = material combination factor.

- $k = \frac{(\sigma_{es})^2 \sin\phi \left[\frac{1}{E_P} + \frac{1}{E_G}\right]}{1.4}$ MPa.
- $k = 0.16 \left[\frac{BHN}{100}\right]^2$ when both gear & pinion are of STEEL

Abrasive : foreign particle
corrosive : chemical action of lubricant
scoring : metal to metal contact.
pitting : due to repeated loading (fatigue)

Q A pair of spur gears having 20 full depth involute teeth is to transmit 20kW. The pinion runs at 300 rpm and speed ratio 3:1. The following data are given. No of teeth on pinion is 15. Service factor = 1; Velocity factor = $\frac{3}{3+v}$, tooth form factor $(y) = \left[0.154 - \frac{0.912}{z}\right]$

Face width, b = 14m; allowable elastic stress for pinion and gear material are 120 MPa and 100 MPa respectively. Surface endurance limit 600MPa. Young's modulus of pinion: 200 GPa; $E_G = 100$ GPa. Design the gear tooth and check for bending and wear failure.

Solⁿ: 1) P = 20kW at 300 rpm.

2) Pressure ratio, $\phi = 20$ F.D.

3) Gear ratio, $G = 3$.

4) No. of teeth on pinion, $z_1 = 15$.

5) Service factor, $C_s = 1$.

6) Velocity factor, $C_v = \frac{3}{3+v}$

7) y = tooth form factor $= 0.154 - \frac{0.912}{z}$.

8) face width $= 14m \Rightarrow \psi = \frac{b}{m} = 14$.

9) Permissible elastic stress for pinion $[\sigma_{b_1}] = 120$ MPa.

" " " " gear $[\sigma_b]_2 = 100$ MPa

10) Surface endurance limit $= K = \sigma_{es} = 600$ MPa.

11) $E_1 = 200$ GPa ; $E_2 = 100$ GPa.

Steps: Subscripts 1 and 2 represents pinion and gear respectively.

1.) T_1 = torque to be transmitted by pinion $= \frac{P \times 60 \times 10^6}{2\pi N_1} = 636619.7724$ N-mm

2) Design torque $[T_1] = C_s \times T_1 = 636619.7724$ N-mm. for pinion.

3) z_2 = no. of teeth on gear $= G z_1 = 45$.

4) Finding the weaker gear [Finding module].

$$[\sigma_{b_1}] Y_1 = 120 \times \left[0.154 - \frac{0.912}{z_1}\right] = 35.1355 \text{ MPa}$$

$$[\sigma_b]_2 Y_2 = 100 \times \left[0.154 - \frac{0.912}{z_2}\right] = 42.0136 \text{ MPa}.$$

$\because [\sigma_b]_1 Y_1 < [\sigma_b]_2 Y_2$ ∴ pinion is the weaker wheel.

$$([\sigma_b] Y)_{WG} = [\sigma_b]_1 Y_1 = 35.135 \text{ MPa}$$

Hence, module should be calculated w.r.t. beam strength of pinion.

$$m \geq 1.26 \sqrt[3]{\frac{T_1}{([\sigma_b]Y)_{WG} \times \psi \times Z_1}} \quad , \quad m \geq 5.5676 \text{ mm} \quad \text{or} \quad \boxed{m = 6 \text{ mm}}$$

5) Dimensions of gear train

$D_1 = mZ_1 = 90\text{mm}$; $D_2 = mZ_2 = 270\text{mm}$, $b = 14m = 84\text{mm}$.

6) F_s = beam strength of WG

$$F_s = ([\sigma_b]Y)_{WG}\, b\, m = 17708.292 \text{ N}.$$

7) F_d = dynamic load:

$$F_t = \frac{2[T_1]}{D_1} = 14147.10605 \text{ N}$$

$$V = \frac{\pi D_1 N_1}{60} = 1.4137 \text{ m/s}.$$

$$C_V = \frac{3}{3+V} = 0.6797.$$

$$F_d = \frac{F_t}{C_V} = 20813.7727 \text{ N}.$$

$\because \underline{F_d > F_s}$

$\therefore$ design is unsafe w.r.t. bending failure.

Hence, new module is calculated using by equating $F_d = F_s$.

8) New module:

$$F_s = F_d = 20813.7727 = ([\sigma_b]Y)_{WG}\, b\, m = (35.1355) \times 14m \times m.$$

$$m = 6.5 \simeq 7\text{mm}.$$

9) Dimensions of gear train (new revised)

$D_1 = mZ_1 = 105\text{mm}$, $D_2 = mZ_2 = 315\text{mm}$, $b = 98\text{ mm}$.

$$F_s = ([\sigma_b]Y)_{WG}\, bm = 24102.953 \text{ N}.$$

10) F_d = dynamic load:

$$F_t = \frac{2[T_1]}{D_1} = 12126.0909 \text{ N}$$

$$V = \frac{\pi D_1 N_1}{60} = 1.6493 \text{ m/s}.$$

$$C_V = \frac{3}{3+V} = 0.6452$$

$$\therefore F_d = \frac{F_t}{C_V} = 18792.7575 \text{ N}$$

$\because F_{d_{revised}} < F_{s_{revised}}$ $\therefore$ design is safe w.r.t. bending ~~strength~~ failure

ii) **Wear strength:** [calculated for pinion only bcoz pinion is weaker w.r.t. wear failure [$\because N_1 > N_2$]]

$F_w = D_1 Q K b.$

$$Q = \frac{2G}{G+1} = \frac{3}{2}.$$

$$K = \frac{(\sigma_{es})^2 \sin\phi}{1\cdot4}\left[\frac{1}{E_1} + \frac{1}{E_2}\right]$$

$$= 1\cdot31922 \text{ MPa.}$$

$$F_w = 105 \times \frac{3}{2} \times k \times 98 = 20362\cdot16923 \text{ N}.$$

$\because F_d < F_w$. $\therefore$ design is safe w.r.t. wear failure.

★ Expression for module [m]:

$$[F_s]_{WG} = ([\sigma_b]Y)_{W.G}.b\,m.$$

$$(F_t)_{max} \le (F_s)_{W.G}.$$

$$\frac{2[T_1]}{D_1} \le ([\sigma_b]Y)_{W.G.}\,\psi(m)(m) \qquad \because b = \psi m.$$

$$\frac{2[T_1]}{mZ_1} \le ([\sigma_b]Y)_{W.G.}\,\psi(m)(m)$$

$$m^3 \ge \frac{2[T_1]}{([\sigma_b]Y)\psi Z_1}$$

$$m \ge 1\cdot26 \sqrt[3]{\frac{[T_1]}{([\sigma_b]Y)\psi Z_1}}.$$

Det. of power transmitted by spur gear when dimensions are known :

I/P data : m, z, ϕ, y, ψ or b, $[\sigma_b]$ = ____ MPa, C_v, C_s and N rpm

1) $F_s = ([\sigma_b]Y)_{WG}\, b\, m$.

$([\sigma_b]Y)_{WG}$ = min. of ($[\sigma_b]_1 Y_1$ and $[\sigma_b]_2 Y_2$).

2) $F_d \le F_s$.

$\left(\frac{F_t}{C_v}\right)$ or $(F_t \times C_v) = F_s$ get F_t = ________ kN

3) Power (P) = $F_t \times V$ = ________ W

4) Rated power = $P \times C_s$ = ________ Watts.

5) $F_w = D_1 Q k b$ = ________

$F_w \geq F_d$

∴ dimns are safe. w.r.t. wear failure.

Q Following data is given for spur gear :

① No. of teeth on pinion, $z_1 = 30$.
② " " " gear, $z_2 = 60$
③ Speed of pinion, $N_1 = 1440$ rpm.
④ Pressure angle, $\phi = 20°$ F.D.
⑤ Module, $m = 3$ mm.
⑥ Face width, $b = 33$ mm.
⑦ Velocity factor, $C_v = \frac{3}{3+v}$
⑧ Lewis form factor, $y = 0.154 - \frac{0.912}{z}$.
⑨ BHN = 200. both are made of steel with $S_{ut} = 560$ MPa.
⑩ Young's modulus, $E = 200$ GPa.
⑪ Service factor = 1.2.

Find rated power on the basis of bending failure if FoS = 1.5 and wear strength.

Soln $[\sigma_b]$ per bending stress $= \frac{560}{1.5} = \frac{1120}{3}$.

∵ Both gear and pinion are made of same material

∴ pinion will be weaker wheel.

$$F_s = \pi \frac{1120}{3}\left[0.154 - \frac{0.912}{30}\right] \times 33 \times 3 = 14351.59949 \text{ N}.$$

$$\frac{2 \times 458366.2361}{16 \times 20.6874 \times 10} \leq m^3$$

$$\therefore \quad m \geq 6.52 \simeq \boxed{m = 7}$$

⑤ Dimensions of the gear train :

$D_1 = mZ_1 = 112\,mm$; $D_2 = mZ_2 = 336\,mm$; $b = 10m = 70\,mm$.

⑥ F_s = beam strength of weaker gear:

$$F_s = ([\sigma_b]Y)_{WG}\, b \cdot m = 10136.826\ N \quad \text{—} ①$$

⑦ F_d = dynamic load:

$$F_t = \frac{2[T_1]}{D_1} = 8185.1113\ N$$

Pitch line velocity ; $V_1 = V_2 = V = \frac{\pi D_1 N_1}{60} = 1.7593\ m/s$

Velocity Factor, $C_v = \frac{4.5}{4.5+V} = 0.7189$.

$$\therefore F_d = \frac{F_t}{C_v} = 11385.1113\ N.$$

$\because F_d > F_s \therefore$ design fails w.r.t. to the beam strength of gear.

⑧ Revising the module: (by equating ~~...~~ $F_d = F_s$).

Using:

$$F_s = ([\sigma_b]Y)_{WG} \times 10m^2 \quad (eq\ ①)$$

$$11385.1113 = 20.6874 \times 10 \times m^2$$

$$m = 7.42$$

$$\boxed{m \simeq 8}$$

⑨ Revising the dimensions of the gear train:

$D_1 = mZ_1 = 128\,mm$; $D_2 = 384\,mm$; $b = 10m = 80\,mm$.

⑩ Revised beam strength of weaker gear

$$F_s = 20.6874 \times 80 \times 8 = 13239.936\ N.$$

⑪ Revised dynamic load:

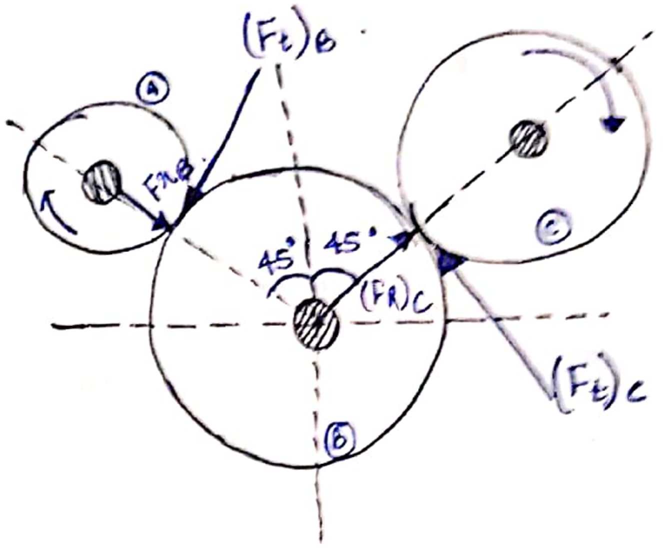
$$F_t = 7161.9724\ N$$

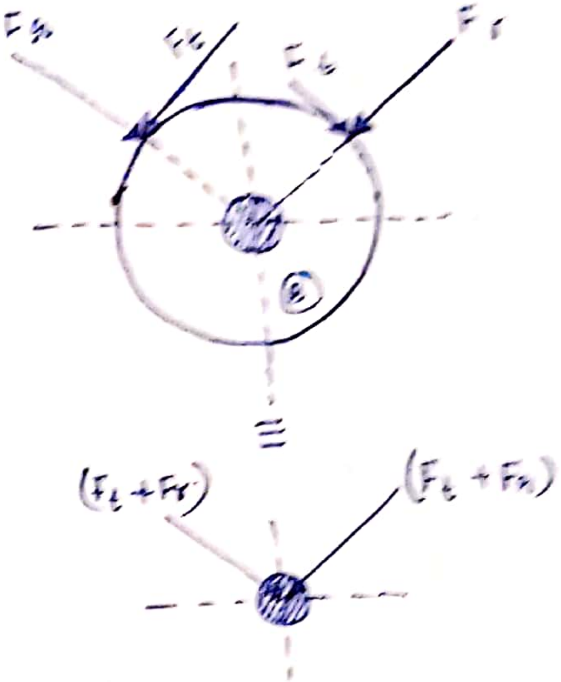
$$V_1 = \frac{\pi D_1 N_1}{60} = 2.0106\ m/s.$$

Velocity factor ; $C_v = \frac{4.5}{4.5+V} = 0.6912$.

$$F_d = \frac{F_t}{C_v} = 10361.9724\ N.$$

$\because F_d < F_s$: design is safe.





$V = 2\,(F_t + F_r)\cos 45°$

$H = 0.$

$(F_R)_2 = (\sqrt{2})\,(F_t + F_r).$

For Gear A.

$$T_A = \frac{60 \times 10^6 \times P}{2\pi N_A} = 47746.48293 \text{ N-mm}$$

$$F_t = \frac{2T_A}{D_A} = 636.6198 \text{ N}$$

$$F_r = F_t \tan\phi = 231.7106 \text{ N}.$$

For Gear B

$$T_B = 0$$

For Gear C

$$\frac{T_A}{T_C} = \frac{Z_A}{Z_C} \Rightarrow T_A = 63661.97724 \text{ N-mm}.$$

Now, Resultant force on idler gear shaft:

$$(F_R)_B = \sqrt{2}\,(F_t + F_r) = 1228 \text{ N}.$$

Q1 Given:

① Pressure angle, $\phi = 14\frac{1}{2}°$ ② Power transmitted, $P = 12 kW$

③ Speed of pinion, $N_1 = 3000 rpm$ ④ Gear ratio $G = 3$

⑤ Permissible elastic stress for pinion $[\sigma_b]_1 = 105$ MPa

⑥ " " " " gear $[\sigma_G]_2 = 60$ MPa

⑦ Surface endurance limit, $\sigma_{es} = 600$ MPa

⑧ No. of teeth on pinion, $z_1 = 16$

⑨ Velocity factor, $C_v = \frac{4.5}{4.5 + V}$

⑩ Form factor, $y = 0.124 - \frac{0.684}{z}$

⑪ Young's modulus for steel pinion, $E_1 = 200$ GPa

" " " CI gear, $E_2 = 100$ GPa

⑫ Assuming for service factor, $C_s = 1.2$.

Let

⑬ Face width, $b = 10m$

i.e. $\psi = 10$

★ design procedure:

Let subscripts 1 and 2 are used for pinion and gear respectively

① T_1 = mean torque to be transmitted by pinion:

$$T_1 = \frac{60 \times 10^6 P}{2\pi N_1} = 381971.8634 \text{ N-mm}$$

② $[T_1]$ = design torque for pinion: $[T_1] = T_1 \times C_s$

$$= 458366.2361 \text{ N-mm}$$

③ No. of teeth on gear: $z_2 = G z_1 = 48$

④ Calculation of module:

↳ Finding the weaker wheel:

Pinion: $[[\sigma_b]_1 Y_1] = 105 \times \pi \times \left(0.124 - \frac{0.684}{z_1}\right) = 26.8017$ MPa

Gear: $([\sigma_b]_2 Y_2) = 60\pi \left[0.124 - \frac{0.684}{z_2}\right] = 20.6874$ MPa

$\because ([\sigma_b]_2 Y_2) < ([\sigma_b]_1 Y_1)$

$\therefore$ Gear is a weaker wheel. Hence design will be based upon gear.

i.e. $([\sigma_b] Y)_{WG} = ([\sigma_b]_2 Y_2) = 20.6874$ MPa

Hence, module should be calculated using the beam strength of gear

$$[F_s]_{WG} = [(\sigma_b) Y]_{WG} \times b \times m$$

$$\frac{2T_1}{m z_1} \leq [(\sigma_b) Y]_{WG} \times 10 m^2$$

$$\frac{2 \times 458366.2361}{16 \times 20.6874 \times 10} \leq m^3$$

$\therefore$ $m \geq 6.52 \simeq \boxed{m = 7}$

⑤ Dimensions of the gear train:

$D_1 = mZ_1 = 112mm$; $D_2 = mZ_2 = 336mm$; $b = 10m = 70mm$.

⑥ F_s = beam strength of weaker gear:

$F_s = ([\sigma_b]Y)_{WG} b \cdot m = 10136.826 \text{ N}$ —①

⑦ F_d = dynamic load:

$$F_t = \frac{2[T_1]}{D_1} = 8185.1113 \text{ N}$$

Pitch line velocity ; $V_1 = V_2 = V = \frac{\pi D_1 N_1}{60} = 1.7593 \text{ m/s}$

Velocity Factor, $C_v = \frac{4.5}{4.5+V} = 0.7189$.

$\therefore F_d = \frac{F_t}{C_v} = 11385.1113 \text{ N}$.

$\because F_d > F_s \therefore$ design fails w.r.t. to the beam strength of gear.

⑧ Revising the module: (by equating ~~Fs~~ $F_d = F_s$).

Using: $F_s = ([\sigma_b]Y)_{WG} \times 10m^2$ (eq ①)

$$11385.1113 = 20.6874 \times 10 \times m^2$$

$$m = 7.42$$

$$\boxed{m \simeq 8}$$

⑨ Revising the dimensions of the gear train:

$D_1 = mZ_1 = 128mm$; $D_2 = 384mm$; $b = 10m = 80mm$.

⑩ Revised beam strength of weaker gear

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⑪ Revised dynamic load:

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Velocity factor ; $C_v = \frac{4.5}{4.5+V} = 0.6912$.

$$F_d = \frac{F_t}{C_v} = 10361.9724 \text{ N}.$$

$\because F_d < F_s$: design is safe.

(12) Calculation of wear load: (always for pinion as pinion is weaker w.r.t. wear failure)

$$K = \frac{\sigma_{es}^2 \sin\phi}{1.4}\left[\frac{1}{E_1} + \frac{1}{E_2}\right] = 0.9858 \text{ MPa}$$

$$Q = \frac{2G}{G+1} = \frac{3}{2}.$$

$$\therefore F_w = D_1 Q K b = 14834.688$$

$\because F_w > F_d \quad \therefore$ design is safe..

Design of Shafts:

Case 1: Under variable load → COMPLETELY REVERSED (strength criterion can be used)
→ FLUCTUATING, → ALTERNATING: These two eq^n are compulsory for these two loads.

(i) Soderberg equation :- (ductile materials)

$$\frac{\sigma_m}{\sigma_{yt}} + \frac{K_f\,\sigma_v}{\sigma_e} = \frac{1}{N} \quad (N_1 = N_2 = N)$$

(ii) Goodman's equation :- (Brittle material)

$$(K_t)\left[\frac{\sigma_m}{\sigma_{ut}}\right] + (K_f)\left[\frac{\sigma_v}{\sigma_e}\right] = \frac{1}{N} \quad (N_1 = N_2 = N)$$

Should be used for design of a component when x-s/c is subjected to either
* Variable axial load
* " bending moment
* " twisting " alone.

where, $\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$ and $\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$;

* $k_f = 1 + q\,(k_t - 1)$
* $\sigma_e = \sigma_e^* K_a K_b K_c$ = endurance limit of a mechanical component (corrected E.L.).

where σ_e^* = E.L. of a std. specimen. → obtained from SN curve
= E.L. under reversed bending.
= $0.5\,S_{ut}$ for Steels.
= $0.4\,S_{ut}$ for CI.

K_a = size factor; K_b = S.F. factor.

K_c = load factor → = 1 for reversed bending.
→ = 0.7 for reversed axial load.
→ = 0.6 for reversed torsion.

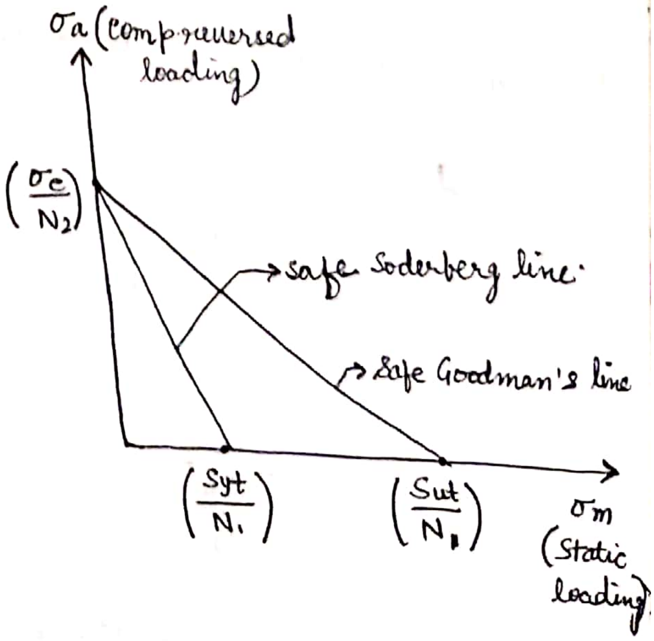
∴ Soderberg eq^n (Actual):

$$(K_t)\frac{\sigma_m}{\left(\frac{S_{yt}}{N_1}\right)} + (K_f)\frac{\sigma_a}{\left(\frac{\sigma_e}{N_2}\right)} = 1.$$

K_t: 1 for ductile.

Goodman's eq^n (Actual):

$$(K_t)\frac{\sigma_m}{\left(\frac{\sigma_{ut}}{N_1}\right)} + (K_f)\frac{\sigma_a}{\left(\frac{\sigma_e}{N_2}\right)} = 1$$



* Soderberg, Goodman and Gerber's eq^n are compulsory for design of component under alternating and fluctuating fatigue load. because failure stress of material is unknown when mean and variable stresses are non zero.

* Strength criterion, Soderberg, Goodman and Gerber's eq^n will give same result under completely reversed fatigue loading. [$\sigma_m = 0$, $\sigma_v = \sigma_{max}$] because failure stress of a material [i.e. σ_{es}] is known. Hence soderberg goodman and gerberg eq^n are optional under completely reversed fatigue loading.

* When $\sigma_m \neq 0$ all the three eq^n will give different results, hence best eq^n should be selected for safe design.

* Soderberg's equation:

↳ When variable A.L. or variable B.M alone :-

$$\frac{1}{N} = \frac{\sigma_m}{\sigma_{yt}} + k_f \frac{\sigma_a}{\sigma_e}. \quad \text{—— (I)}$$

↳ When variable TM acts alone :-

$$\frac{1}{N} = \frac{\tau_m}{\tau_{ys}} + k_f \frac{\tau_a}{\tau_e} \quad \text{—— (II)}$$

↳ When more than one variable load is acting ;

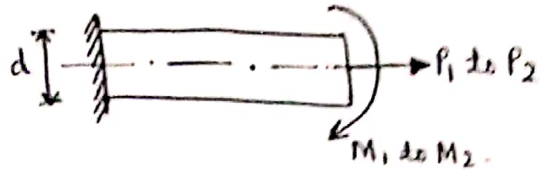
$$\frac{S_{yt}}{N} = \sigma_m + K_f \frac{\sigma_a S_{yt}}{\sigma_e} \qquad \text{From eq (I)}$$

$$\therefore \boxed{\sigma_{eq} = \sigma_m + K_f \frac{\sigma_a \sigma_{yt}}{\sigma_e}} \longrightarrow \text{(III)}$$

For T.M :

$$\frac{S_{ys} \text{ or } \tau_{ys}}{N} = \tau_m + \frac{K_f \tau_a \tau_{ys}}{\tau_e}.$$

$$\boxed{\tau_{eq} = \tau_m + \frac{K_f \tau_a \tau_{ys}}{\tau_e}} \quad \text{—— (IV)}$$

① 

From eq III :

$$(\sigma_{eq})_a = \frac{4P_m}{\pi d^2} + K_f \frac{4P_a}{\pi d^2} \cdot \frac{\sigma_{yt}}{\sigma_e}.$$

$$(\sigma_{eq})_a = \frac{X}{d^2} \text{ MPa} \quad \text{—— (A)}$$

From eq III:

$$(\sigma_{eq})_b = \frac{32 M_m}{\pi d^3} + K_f \left[\frac{32 M_a}{\pi d^3}\right]\left[\frac{\sigma_{yt}}{\sigma_e}\right].$$

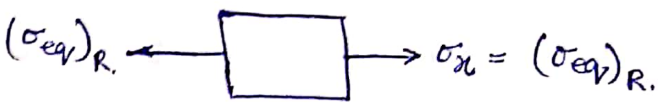
$$(\sigma_{eq})_b = \frac{Y}{d^3} \quad \text{—— (B)}$$

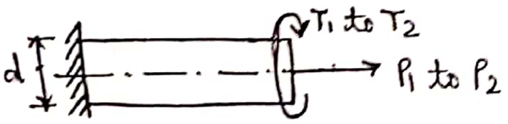
$$\therefore (\sigma_{eq})_R = (\sigma_{eq})_a + (\sigma_{eq})_b = \frac{X}{d^2} + \frac{Y}{d^3} \quad \text{—— (C)}$$

$$(\sigma_{max})_{ind} \le \sigma_{per}.$$

$$\frac{X}{d^2} + \frac{Y}{d^3} \le \frac{S_{yt}}{N}.$$

$$\boxed{d \ge \text{——} \text{ mm}}$$



② 

From eq III : $$(\sigma_{eq})_a = \frac{4P_m}{\pi d^2} + K_f \frac{4P_a}{\pi d^2} \cdot \frac{\sigma_{yt}}{\sigma_e}.$$

$$(\sigma_{eq})_a = \frac{X}{d^2} \text{ MPa} \quad \text{—— (A)}$$

From eq IV :

$$\tau_{eq} = \frac{16 \tau_m}{\pi d^3} + K_f \frac{16 \tau_a}{\pi d^3} \left(\frac{\tau_{ys}}{\tau_e}\right)$$

$\frac{S_{yt}}{2}$ MSST

$\frac{S_{yt}}{\sqrt{3}}$ MDET

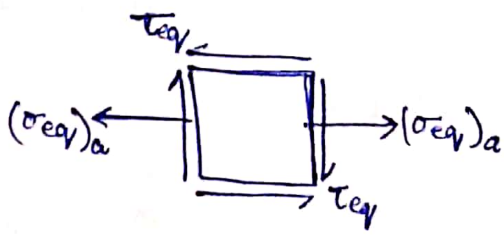
$$\tau_{eq} = \frac{Z}{d^3} \text{ MPa} \quad \text{—— (B)}.$$

by using MSST:

$$(\sigma_t)_{per} = \sqrt{\sigma_x^2 + 4\tau_{xy}^2}$$

$$\tau_{per} = \frac{1}{2}\sqrt{\sigma_x^2 + 4\tau_{xy}^2}$$

$$\frac{S_{ys}}{N} = \frac{1}{2}\sqrt{(\sigma_{eq})_a^2 + 4\tau_{eq}^2} \quad ; \; d = \text{——} \text{ mm}.$$



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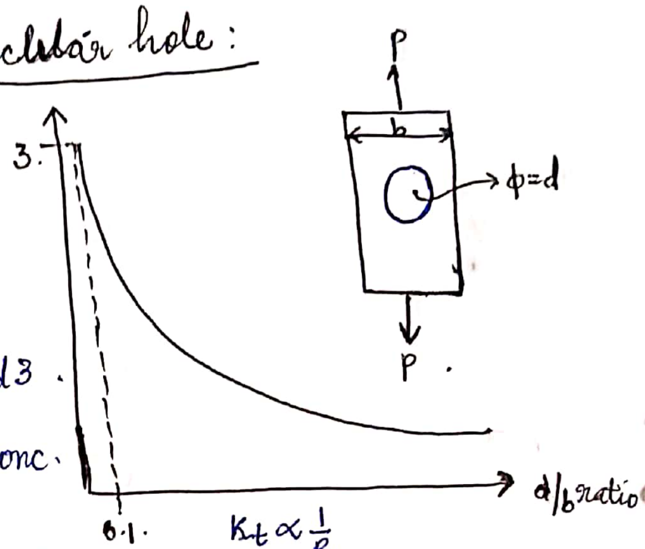
by using MDET:

$$(\sigma_t)_{per} = \sqrt{\sigma_x^2 + 3\tau_{xy}^2}$$

$$\tau_{per} = \frac{1}{\sqrt{3}}\sqrt{\sigma_x^2 + 3\tau_{xy}^2}$$

$$\frac{S_{yc}}{N} = \frac{1}{\sqrt{3}}\sqrt{(\sigma_{eq})_a^2 + 3(\tau_{eq})^2}$$

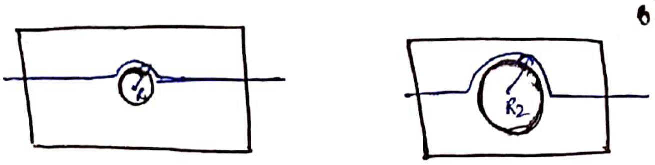
* Stress concentration factor for circular hole:



* For circular hole, max stress conc. factor = 3. its actual value depends on $\frac{d}{b}$ ratio

Hence; stress conc. factor lies b/w 1 and 3.

* Smaller hole will have higher stress conc. factor.



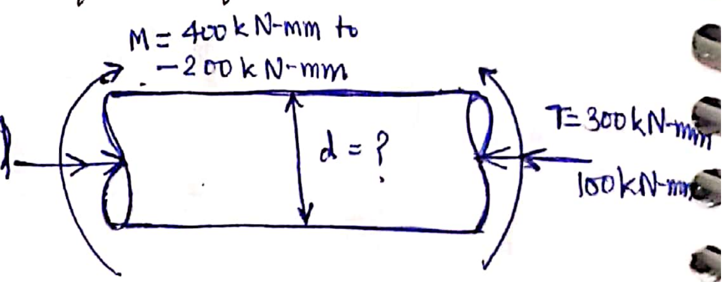
$K_t \propto \frac{1}{R}$.

$R_1 < R_2$.

Stress conc. in first case is more.

Q A hot rolled steel shaft subjected to torsional moment varies from 300 kN-mm (CW) to 100 kN-mm (ACW) and bending moment at critical crossection varies from 400 kN-mm to −200 kNmm. the shaft is of uniform diameter and no keyway is present at the critical x-s/c. Determine diameter of shaft. by taking FoS = 1.5. assume ultimate, yield and design stress at 560 MPa, 420 MPa and 280 MPa. Modification factor = 0.62 ; size correction factor = 0.85 load factor for bending = 1. load factor for torsion = 0.58.

Sol^n Given:



$K_t = K_f = 1$ [because of no discontinuity]

$S_{yt} = \sigma_{yt} = 420$ MPa.

$S_{ut} = \sigma_{ut} = 560$ MPa.

design stress $(\sigma_e^*) = 280$ MPa.

Size factor, $K_a = 0.85$

Surface finish correction factor $= k_b = 0.62$.
load factor for bending $k_c = 1$
load factor for torsion $k_c = 0.58$
F.O.S. $N = 1.5$.

① **Assuming variable bending moment acting alone :-**

Soderberg eqⁿ is used because of ductile material (hot rolled steel).

$M_{max} = 400$ kN-mm ; $M_{min} = -200$ kN-mm.

$M_m = \dfrac{M_{max} + M_{min}}{2} = 100$ kN-mm ; $M_a = \dfrac{M_{max} - M_{min}}{2} = 300$ kN-mm.

$\sigma_m = \dfrac{32 M_m}{\pi d^3} = \dfrac{3200000}{\pi d^3}$ MPa ; $\sigma_a = \dfrac{32 M_a}{\pi d^3} = \dfrac{96 \times 10^5}{\pi d^3}$ MPa.

$\sigma_e = \sigma_e^* K_a K_b K_c = 280 \times 0.85 \times 0.62 \times 1 = 147.56$ MPa.

Hence, from soderbergs eqn:

$$\sigma_{eq} = \sigma_m + K_f \frac{\sigma_a \sigma_{yt}}{\sigma_e} = \frac{30.5245 \times 10^6}{\pi d^3} \text{ MPa} \quad \text{—①}$$

② **Assuming variable twisting moment alone :-**

$T_{max} = 300$ kN-mm ; $T_{min} = -100$ kN-mm.

$T_m = \dfrac{T_{max} + T_{min}}{2} = 100$ kN-mm ; $T_a = \dfrac{T_{max} - T_{min}}{2} = 200$ kN-mm.

$\tau_m = \dfrac{16 T_m}{\pi d^3} = \dfrac{16 \times 10^5}{\pi d^3}$ MPa ; $\tau_a = \dfrac{16 T_a}{\pi d^3} = \dfrac{32 \times 10^5}{\pi d^3}$ MPa.

$\tau_e = \sigma_e^* (K_a)(K_b)(K_c) = 280 \times 0.85 \times 0.62 \times 0.58 = 85.5848$ MPa.

Using soderberg's eqn:.

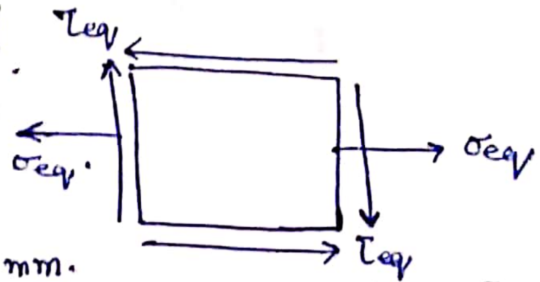
$$\tau_{eq} = \tau_m + K_f \frac{\tau_a \tau_{ys}}{\tau_e}$$

[Ussing MSST ; $\tau_{ys} = \dfrac{S_{yt}}{2}$].
$\tau_{ys} = \dfrac{420}{2}$.

$$\tau_{eq} = \frac{9.4519 \times 10^6}{\pi d^3} \text{ MPa}$$

③ **Diameter of shaft : (d)**

diameter is determined by using MSST because of biaxial state of stress and ductile material.

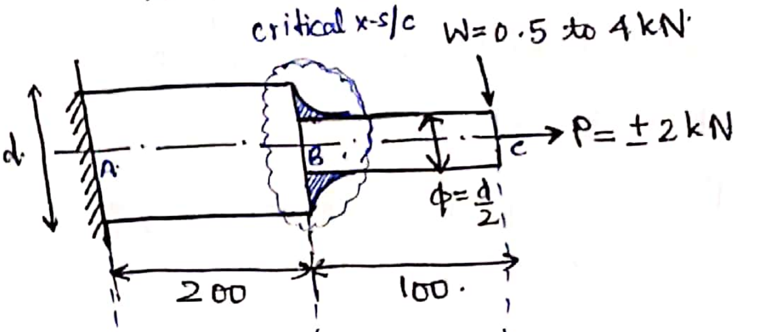


$$\tau_{per} = \sqrt{(\sigma_{eq})^2 + 4(\tau_{eq})^2}$$

$$\frac{420}{2 \times 1.5} = \frac{35.9040 \times 10^6}{\pi d^3 \times 2}$$

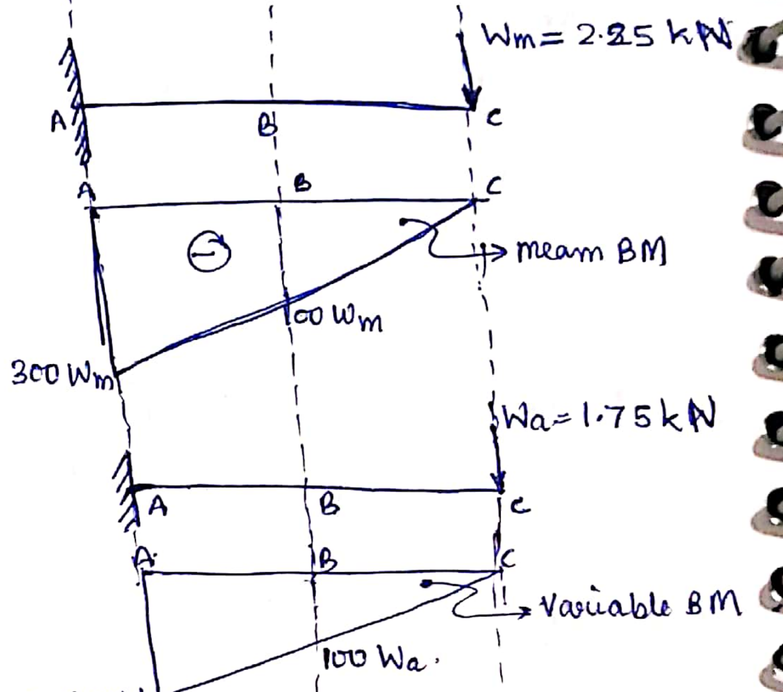
$d = 34.433798$ mm.
$d \simeq 35$ mm. [Standard dia in series of 5]

Q For the stepped bar as shown determine for an infinite life, Assume $K_a = 0.9$, $K_b = 0.8$, $K_t = 1.5$, $q = 0.9$; $S_{yt} = 250\,MPa$, $S_{ut} = 400$
$F.o.S = 2$



$$\frac{[(\sigma_b)_{max}]_B}{[(\sigma_b)_{max}]_A} = \frac{(M_{max})_B}{(M_{max})_A}\cdot\left[\frac{d_A}{d_B}\right]^3$$
$$= \frac{1}{3}[8] = \frac{8}{3}.$$

$$\frac{(\sigma_{axial})_B}{(\sigma_{axial})_A} = \left(\frac{d_A}{d_B}\right)^2 = 4.$$

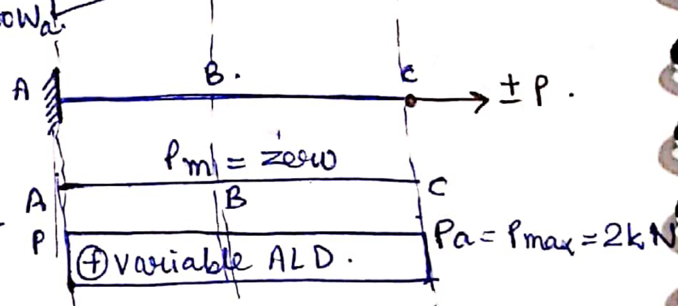


Hence, critical x-s/c is the x-s/c where dis continuity present. i.e. at B.
[∵ $(\sigma_b)_B = \frac{8}{3}(\sigma_b)_A$ and $(\sigma_a)_B = 4(\sigma_a)_A$]

* Stress analysis should be carried at x-s/c 'B'.

① When 'W' is acting alone:

$$\sigma_m = \frac{32 M_m}{\pi d^3} = \frac{32\times 100\times 2.25\times 10^3}{\pi\left(\frac{d}{2}\right)^3} = \frac{57.6\times 10^6}{\pi d^3}$$



$$\sigma_a = \frac{32 M_a}{\pi d^3} = \frac{32\times 100\times 1.75\times 10^3}{\pi\left(\frac{d}{2}\right)^3} = \frac{44.8\times 10^6}{\pi d^3}.$$

$K_f = 1 + q(K_t - 1) = 1.45.$

$\sigma_e = (\sigma_e^*)(K_a\cdot K_b\cdot K_c) = 144\ MPa.$

$\sigma_e^* = 50\%$ of S_{ut}.
$= 200.$

by using Soderberg's eqn:

$$(\sigma_{eq})_b = \sigma_m + K_f\frac{\sigma_a\sigma_{yt}}{\sigma_e} = \frac{17.2629\times 10^6}{\pi d^3}\ MPa$$

When completely reversed axial load (P) acts:

$\sigma_m = 0$
$$\sigma_a = \sigma_{max} = \frac{4P_a}{\pi d_B^3} = \frac{32\times 10^3}{\pi d^2}$$

$$(\sigma_{eq})_a = 0 + K_f \frac{\sigma_a \sigma_{yt}}{\sigma_e} = \frac{11.6599 \times 10^3}{\pi d^2} \text{ MPa}.$$

$$\sigma_e = \sigma_e^* K_a K_b K_c = 100.8 \text{ MPa} \qquad \underline{K_c = 0.7}.$$

∴ <u>diameter of bar (d):</u>

$$(\sigma_{max})_{ind} \leq \sigma_{per}.$$

$$\frac{17.2629 \times 10^6}{\pi d^3} + \frac{11.6599 \times 10^3}{\pi d^2} \leq \frac{250}{2}.$$

$$d \geq 76.994 \text{ mm}$$

$$d = 77 \text{ mm}.$$

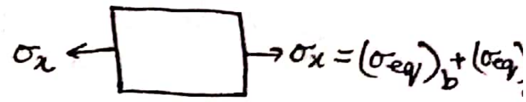


Fig. State of stress at critical plane (i.e. top fibre of critical x-s/c (B)).

Q The non-rotating shaft is subjected to load P varying 4kN to 12kN. S_{ut} = 600 MPa; S_{yt} = 350 MPa; Endurance limit = 300 MPa; size factor = 0.8 ~~MPa~~, surface finish factor = 0.85, q = 0.9; FoS = 3.5, K_t = 1.8. Find the diameter D using Goodman's eq^n:

$$\frac{(\sigma_b)_{B+D}}{(\sigma_b)_C} = \left[\frac{M_B}{M_C}\right]\left[\frac{d_C}{d_B}\right]^3$$

<u>mean σ_b:-</u>

$$\frac{(\sigma_b)_{B+D}}{(\sigma_b)_C} = \frac{240}{240} \times 2^3 = 4.8.$$

<u>Variable σ_b:-</u>

$$\frac{(\sigma_b)_{B+D}}{(\sigma_b)_C} = \frac{120}{120} \times 2^3 = 4.8$$

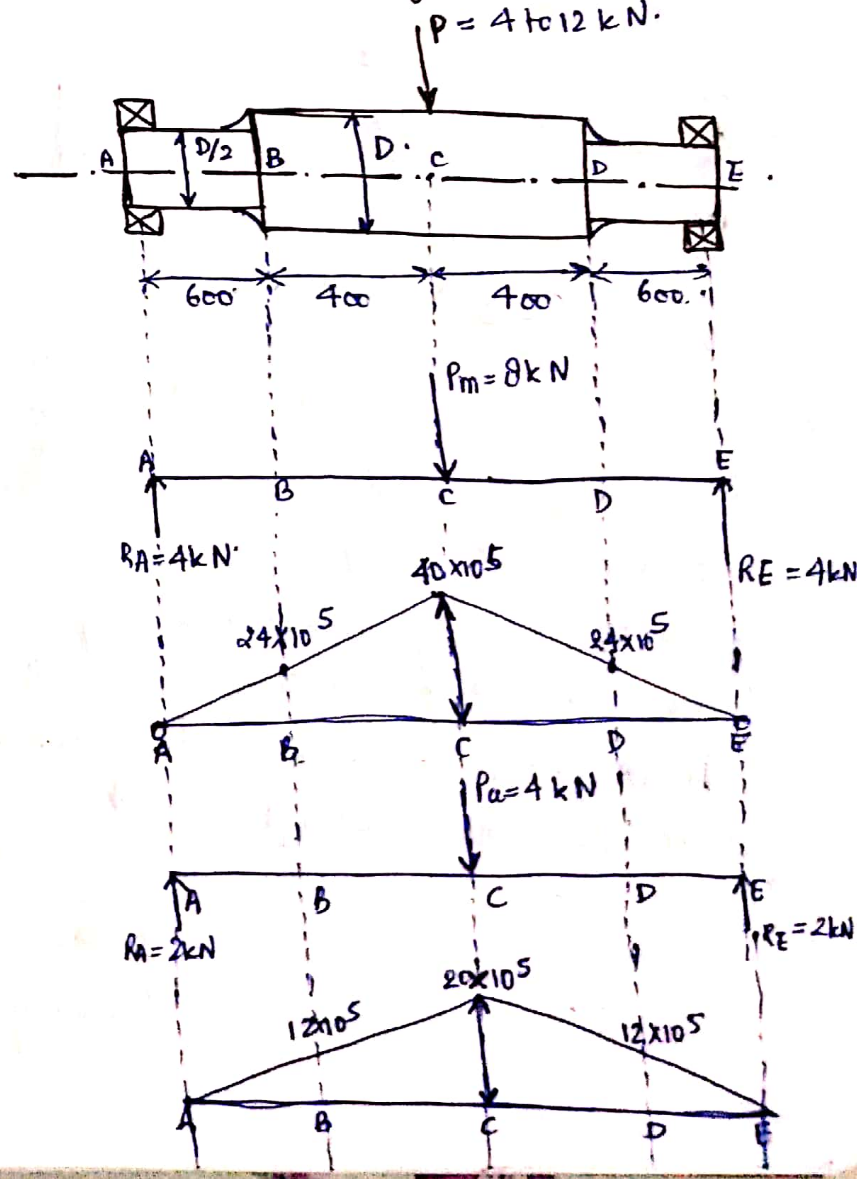
At critical x-s/c:

$$\sigma_m = \frac{32 M_m}{\pi d^3} = \frac{32 \times 24 \times 10^5 \times 8}{\pi D^3}$$

$$= \frac{614.4 \times 10^6}{\pi D^3}.$$

$$\sigma_v = \frac{32 M_a}{\pi d^3} = \frac{32 \times 12 \times 10^5 \times 8}{\pi D^3}$$

$$= \frac{307.2 \times 10^6}{\pi D^3}.$$



$K_f = 1 + q(K_t - 1) = 1.72.$

$\sigma_e = (\sigma_e^*)(K_a)(K_b)(K_c) = 204 \text{ MPa}.$

Using Goodman's eq^n:

$$\frac{1}{N} = K_t\left(\frac{\sigma_m}{\sigma_{ut}}\right) + K_f\left(\frac{\sigma_a}{\sigma_e}\right).$$

$$\frac{1.8432\times10^6}{\pi d^3} + \frac{2.5901\times10^6}{\pi d^3} = \frac{1}{3.5}$$

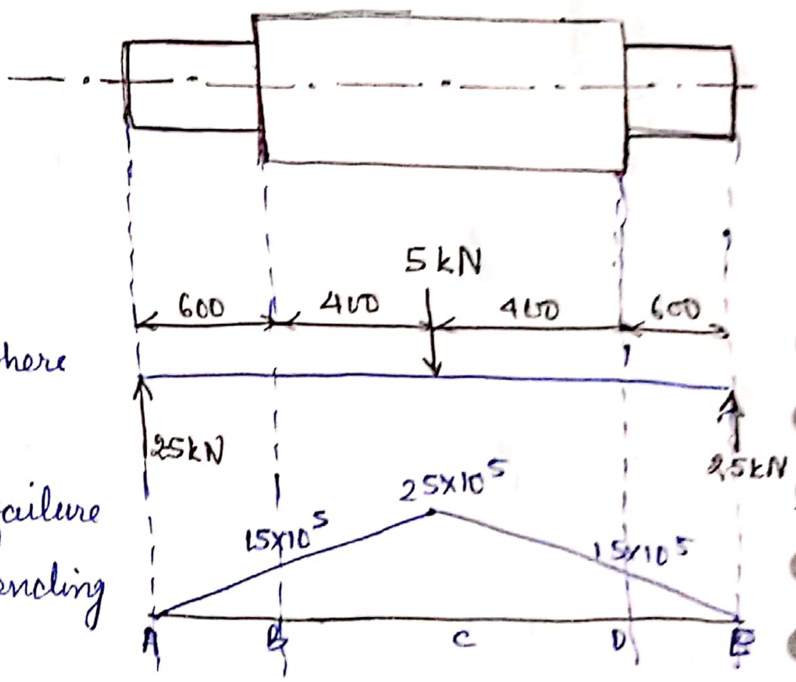
$$d = 170.3 \text{ mm}$$

$$d \simeq 175 \text{ mm}.$$

Q For the rotating shaft as shown in fig: Determine the life of shaft.
$S_{ut} = 500$ MPa; endurance limit of shaft is 200 MPa,
K_f = fatigue stress conc. factor = 2.

$$\frac{(\sigma_b)_{B\&D}}{(\sigma_b)_C} = \frac{M_B}{M_C} \times \left(\frac{d_C}{d_B}\right)^3$$

$$\frac{(\sigma_b)_{B\&D}}{(\sigma_b)_C} = \frac{15}{25} \times 2^3 = 4.8$$



* Critical x-s/c are B & D (i.e. where fillets are provided)
* Shaft is subjected to fatigue failure because completely reversed bending stresses are developed at the critical points (i.e. on bottom fibres) on the critical x-s/c due to rotation of shaft.

$\Downarrow$

fatigue loading.
failure is under tension only.
(rarely in compression)

Critical cross-section :

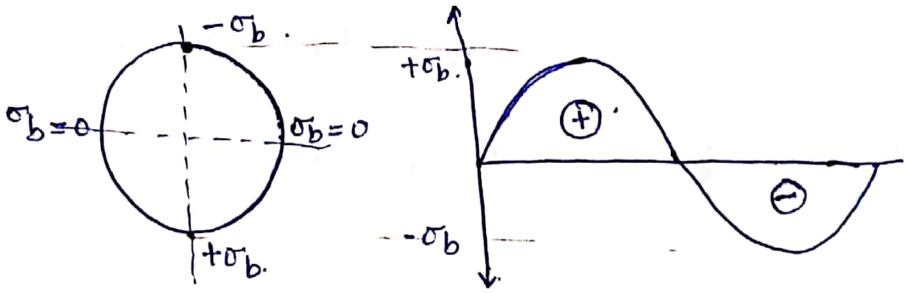
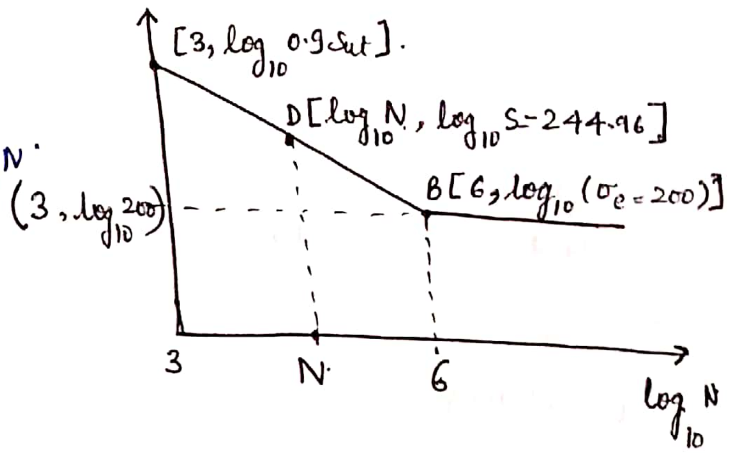


Fig. Stress cycle for comp. reversed bending stress.

$$(\sigma_{max}) = K_f (\sigma_b)_{max} = 2 \times \frac{32M}{\pi(50)^3} = 244.96 \text{ MPa} > \sigma_e.$$

Hence, shaft will have finite life.

$$\frac{\log_{10}(0.9 \times 500) - \log_{10}(200)}{\log_{10}(244.46) - \log_{10}(200)} = \frac{6-3}{6-\log_{10} N}$$



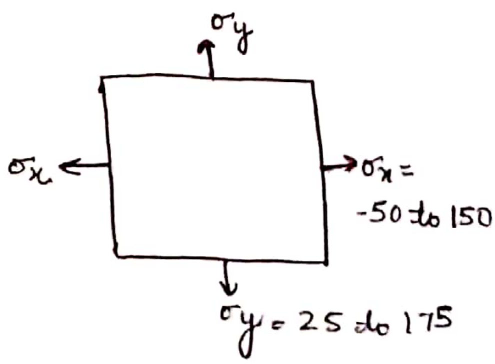
N = 180868.1 cycles.

N = 180869 cycles.

* If diameter is 75 mm instead of 50mm Shaft will have infinite life because, the max stress is less than σ_e.

i.e. $\sigma_{max} = 72.43 < 200$ MPa.

Q A m/c component is subjected to biaxial state of stress as shown. Determine FoS by using MDET.. Assume, Syt = 250 MPa., Sut = 500 MPa, $K_t = 1.85$; $q = 0.9$; corrected endurance limit = 200 MPa.



$$(\sigma_x)_m = \frac{150-50}{2} = 50 \text{ MPa}.$$

$$(\sigma_x)_a = \frac{150+50}{2} = 100 \text{ MPa}.$$

$$K_f = 1 + q(K_t - 1) = 1.765.$$

$$\sigma_e = \sigma_e' * K_a K_b K_c = 200 \text{ MPa}.$$

$$(\sigma_x)_{eq} = \sigma_m + K_f \frac{\sigma_a \sigma_{yt}}{\sigma_e} = 270.625 \text{ MPa}.$$

$(\sigma_y)_{mean} = 100\,MPa; \quad (\sigma_y)_a = 75\,MPa.$

$$(\sigma_y)_{eq} = 100 + 1.765 \times \frac{75 \times 250}{200} = 265.46875\ MPa.$$

$\sigma_1 = 270.625$ [larger of $(\sigma_x)_{eq}$ & $(\sigma_y)_{eq}$].

$\sigma_2 = 265.46875\ MPa.$

by using MDET

$$\sigma_1^2 + \sigma_2^2 - \sigma_1\sigma_2 \leq \left(\frac{S_{yt}}{FoS}\right)^2.$$

$$\underline{FoS = 0.9325}$$

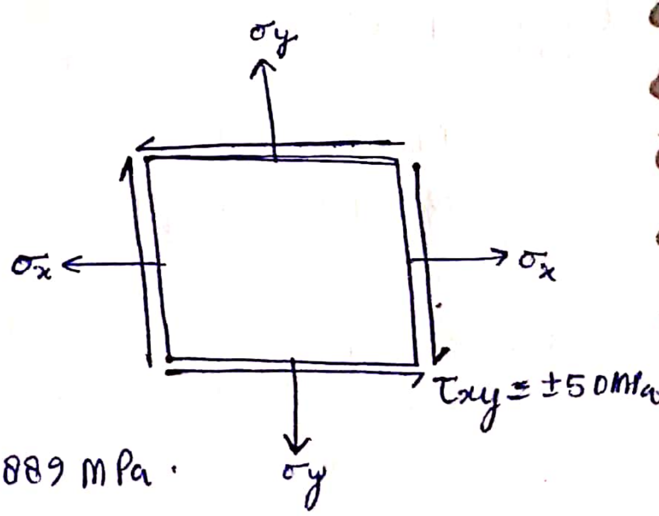
<u>Q</u> If τ_{xy} is also given,..

$\tau_m = 0$

$\tau_a = 50\,MPa.$

$$\tau_{eq} = \tau_m + K_f \frac{\tau_a \tau_{sy}}{\tau_e} \quad \tau_e \to \sigma_e \times K_c$$

$$= 0 + \frac{1.765 \times 50 \times 250}{200 \times \sqrt{3}} = 63.6889\ MPa.$$



Q $S_{ut} = 400$ MPa.

$S_{yt} = 250$ MPa.

$K_f = 2$; $N = 1.5$; $K_a = 0.85$; $K_b = 0.7$